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# DESIGN AND EVALUATION OF A FLEXIBLE, THRUST-MEASURING TEST-STAND MOUNT

by

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ABSTRACT. The report discusses the design and evaluation of the flexure—cell, a flanged column with strain gages attached. Tables, graphs, and equations show the characteristics of the flexure—cell. Results show that this flexible test-stand mount is cheap, hence expendable, and capable of thrust measurements of fairly high accuracy.

U.S. NAVAL ORDNANCE TEST STATION

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U. S. NAVAL ORDNANCE TEST STATION

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## FOREWORD

This report describes a radically new approach to the design of a six-component static-firing test stand for use in hazardous tests where the possibility of complete destruction of the stand is high. A major objective of the development was to provide an inexpensive test stand so that the loss would not be great in case of a catastrophic malfunction.

The design and fabrication of the test stand were carried out on a short time scale and no optimization of components was possible. The general approach proved to be successful, however, and the information in this report should be useful to others engaged in the design of stands for high-hazard tests.

The work described was performed during the summer of 1960, and culminated in a full-scale Polaris test firing in September 1960. The work was supported by Bureau of Ordnance Task Assignment SP 71401.

This report was reviewed for technical accuracy by Foy McCullough, Jr.

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# INTRODUCTION

The number of rocket-motor tests involving possible or deliberate catastrophic failures has increased. Such tests may result in the destruction of the test stand and its fixtures, which means a highly undesirable expenditure of time and money.

The six-component test stand is probably the most expensive test stand in use today. A large part of its cost is in the flexure pivots and load cells that provide freedom of movement for the test stand and measure the thrust developed by the motor, respectively. Commercial flexure pivots and load cells have heretofore been separate units and, although accurate, are expensive.

In mid-1960, the U. S. Naval Ordnance Test Station (NOTS) was asked to design and fabricate a six-component test stand for use in a test that might result in an engine failure of a violent nature. It was deemed desirable to design and manufacture a single-unit flexure pivot—load cell. This item, hereafter called a flexure—cell, consisted of a cylindrical column flexure on which a balanced pattern of strain gages was arranged. The narrow column allowed relatively free test-stand motion, and the strain gages provided a means of measuring tensile or compressive loads free of bending effects. The result greatly reduced the number of items necessary for a six-component test stand as well as the over-all cost of the stand. Figure 1 shows the location of the flexure—cells on the stand; Fig. 2 illustrates the location of commercial items when used on the same test stand.

It should be noted that a conventional Baldwin-Lima-Hamilton Corp. 50,000-pound universal load cell was used to measure axial thrust rather than a flexure—cell. NOTS-built column flexures were used in conjunction with this one commercial load cell, however.

Table 1 gives a cost analysis of the most inexpensive commercial flexure—load cell combination compared to the flexure—cell arrangement actually used. The number of commercial flexure pivots could be cut in half by the use of universal flexures instead of two modular flexures in series as listed. The use of universal flexures, however, greatly increases the over-all cost of the test stand.

TABLE 1. COST OF COMMERCIAL FLEXURE PIVOTS AND LOAD CELLS COMPARED WITH COST OF FLEXURE—CELLS USED ON A SIX-COMPOUND TEST STAND

| Six-component test stand | Cost |
|---|---|
| Using commercial items | |
| 10,000-lb universal load cell (2) | $ 750 |
| 15,000-lb compression load cell (4) | 1,320 |
| 15,000-lb modular flexure pivot (24) | 4,800 |
| 50,000-lb modular flexure pivot (4) | 1,400 |
| 50,000-lb universal load cell | 770 |
| Total | $9,040 |
| Using flexure—cells | |
| 6 flexure—cells, 2 column flexures | $1,650 |
| 50 strain gages | 150 |
| Assembly and calibration | 600 |
| 50,000-lb universal load cell | 770 |
| Total | $3,170 |

# THE FLEXURE-CELL

It was necessary to design and manufacture the flexure-cell, and to determine its effectiveness as both a flexure pivot and a load cell.

## DESIGN

The design of the flexure-cell was based on the established column formula with the arbitrary choice of a 12-inch length.

$$P = \frac{\pi^2 EI}{L^2}$$

where $P$ = design load per flexure-cell, in pounds
$L$ = effective length of column, taken as actual length of 12 inches to provide safety factor
$I$ = $(\pi/4)r^4$, moment of inertia of column
$E$ = $30 \times 10^6$ psi, assumed modulus of elasticity of steel

Table 2 shows the results of the use of this equation.

TABLE 2. VALUES FOR FLEXURE-CELLS AND COLUMN FLEXURES CALCULATED FROM BASIC DESIGN EQUATION

| Item | Design load, lb | Calculated radius of column, in. | Radius of column used, in. | Moment of inertia, $in^4$ |
|---|---|---|---|---|
| Side flexure-cell | 10,000 | 0.28 | 0.30 | 0.00636 |
| Bottom flexure-cell | 15,000 | 0.311 | 0.375 | 0.0156 |
| Axial column flexure | 50,000 | 0.42 | 0.50 | 0.0442 |

## FABRICATION

Figures 3 and 4 show the two types of flexure-cells fabricated and tested, and the axial column flexure used to support the commercial load cell. Each of these items was manufactured from S.A.E. 4130 four-inch-diameter steel bar stock.

Each billet was ultrasonically inspected for flaws, then machined to approximately 0.1 inch oversize on all dimensions. The rough-cut pieces were heat treated to 140,000-psi minimum yield strength (Rockwell C hardness of 38 to 40). Each piece was ultrasonically tested again for flaws before the final machining and drilling.

The strain gages used on the flexure-cells were Tatnall Metalfilm foil strain gages, type C6-121. Eight of these gages were installed in the center section of each flexure-cell with each diametrically opposed pair constituting one leg of the bridge circuit, as shown in Fig. 5. This configuration eliminated bending effects that would have otherwise affected the readings taken from the gages. The final gage pattern allowed only tensile or compressive loads to be measured.

To protect the gages and their leads from dust, moisture, and rough handling, the gages were enclosed in a protective covering (177 g of General Electric Co. silicone rubber compound RTV-60 and 2.66 g of General Electric Co. Thermolite 12). After application, the rubber-like coating was exposed to infrared heating lamps, which reduce the curing time to 3 or 4 hours.

Flexure-cells 1 and 3 had been calibrated before the RTV-60 application, and they were recalibrated with no change in characteristics. Flexure-cells 2, 4, 5, and 6, however, showed an electrical imbalance in the bridge circuit from 0.4 to 0.6 volt, which rendered them useless.

No conclusions were reached as to what had caused the imbalance in the gage circuit. One theory postulates that the RTV-60 and/or its catalyst had somehow acted on the gages or wiring to destroy their effectiveness.

The damaged flexure-cells were stripped of their gages and new gages were applied. In the reapplication of gages it was necessary to use larger gages, Tatnall Metalfilm type C6-141, on two of the flexure-cells because of a shortage of the smaller type C6-121. Use of the C6-141 gages resulted in no apparent disadvantages.

Protection was provided the gages and wiring by enclosing them in a loose cylinder of cardboard bound with electrical tape.

## EFFECTIVENESS AS A FLEXIBLE MOUNT

In order to be used as a flexure, the flexure-cell had to contribute a relatively small resistance to test-stand motion.

By use of classical beam theory, equations were derived for the theoretical restraint to test-stand motion contributed by each flexure-cell and by the axial column flexures (see Appendix A).

The theoretical restraint in pounds per 0.01 inch of test-stand motion perpendicular to the axis of each flexure-cell or column flexure follows:

| | |
|---|---|
| Side flexure-cell .................................................. | 17 |
| Bottom flexure-cell ................................................ | 42 |
| Axial column flexure and load cell assembly ....... | 4.7 |

It should be noted that the theoretical results shown above were based on a mean column length and were not determined by a completely rigorous mathematical proof.

It is seen that a sizable amount of restraint is provided by these flexure-cells; however, this is a repeatable error. For the specific test for which these were designed, however, these inherent inaccuracies in thrust measurement were tolerable (see Appendix B).

In addition to flexibility, it was also of prime importance that the flexure-cells be both strong and stable, especially with offset. Offset is defined as the lateral deflection of one end of the flexure-cell or of the column flexure and load cell assembly with respect to the opposite end.

Laboratory tests on the flexure-cells showed them to be extremely strong and stable even at high values of offset. Figure 6 shows the ultimate strength of a side flexure-cell tested in compression with 0.01-inch offset.

Figure 7 shows the total vertical deflection between end-flange faces of a side flexure-cell as a function of compressive load. The change in slope of the plots, with and without offset, is due to the shape that the piece assumes when its ends are offset. With no offset the compressive deflection in the flexure-cell is the same as the deflection between its flange faces. With offset, however, the total compressive deformation in the piece is not the same as the vertical deflection between flange faces because of the S-shape of the flexure-cell. Figure 8 illustrates this fact.

Figure 7 shows that the flexure-cell did withstand design load with as much as 0.25-inch offset.

## EFFECTIVENESS AS A LOAD-MEASURING DEVICE

Inasmuch as the flexure-cell must provide not only a reasonable flexibility but also an accurate means of measuring the tensile or compressive loads imposed on the member, the following requirements were deemed necessary:

1. The data (load versus strain-indicator reading) should be linear from no load to design load.
2. The amount of offset should not affect the data.

3. The flexure—cell should be sensitive to incremental loading at all load levels.
4. The data should remain constant with repeated loading.

Laboratory tests showed that the linearity of applied load versus strain-indicator reading was very good (see Fig. 9—20). It should be noted that the strain-indicator readings did not give actual deflection in μin because of the method of testing. Instead, these readings are merely a strain-indicator index of load conditions.

Offset had little or no effect on the readings when compared with readings at no offset (see Table 3).

TABLE 3. COMPARISON OF STRAIN-INDICATOR READINGS AT NO OFFSET AND AT 0.1-IN. OFFSET ON BOTTOM FLEXURE—CELL.

| Compressive load, lb | Reading with no offset | Reading with 0.1-in. offset |
|---|---|---|
| 0 | 4,645 | 4,645 |
| 1,000 | 4,460 | 4,455 |
| 2,000 | 4,270 | 4,260 |
| 3,000 | 4,080 | 4,070 |
| 4,000 | 3,885 | 3,880 |
| 5,000 | 3,690 | 3,685 |
| 6,000 | 3,500 | 3,495 |
| 7,000 | 3,305 | 3,300 |
| 8,000 | 3,115 | 3,115 |
| 9,000 | 2,930 | 2,925 |
| 10,000 | 2,740 | 2,735 |
| 12,000 | 2,360 | 2,355 |
| 14,000 | 1,975 | 1,975 |
| 15,000 | 1,795 | 1,795 |

Sensitivity tests on side flexure—cell 2 showed that at all levels of loading the smallest increment of load detectable on a Baldwin Strain Indicator was 25 pounds.

Repeatability tests on a side flexure—cell showed no change in strain-indicator reading during ten loading cycles (see Table 4).

TABLE 4. STRAIN-INDICATOR READINGS UNDER REPEATED LOADS ON BOTTOM FLEXURE—CELL

| Cycle | Strain-indicator reading | | | | | |
|---|---|---|---|---|---|---|
| | No load | 2,000 lb | 4,000 lb | 6,000 lb | 8,000 lb* | 10,000 lb* |
| 1 | 2,250 | 1,660 | 1,070 | 475 | 9,340 | 8,750 |
| 2 | 2,250 | 1,655 | 1,065 | 475 | 9,340 | 8,750 |
| 3 | 2,250 | 1,655 | 1,065 | 475 | 9,340 | 8,750 |
| 4 | 2,250 | 1,655 | 1,070 | 475 | 9,340 | 8,750 |
| 5 | 2,250 | 1,655 | 1,065 | 480 | 9,340 | 8,750 |
| 6 | 2,250 | 1,655 | 1,065 | 480 | 9,340 | 8,750 |
| 7 | 2,250 | 1,655 | 1,070 | 480 | 9,340 | 8,750 |
| 8 | 2,250 | 1,660 | 1,070 | 480 | 9,340 | 8,750 |
| 9 | 2,250 | 1,660 | 1,070 | 480 | 9,340 | 8,750 |
| 10 | 2,250 | 1,660 | 1,070 | 480 | 9,340 | 8,750 |

* Range of Strain Indicator changed to Range Extender B.

## USE OF THE FLEXURE-CELL TEST STAND

In September 1960, a test stand utilizing flexure–cells was used for a destruct test of a full-scale Allegany Ballistics Laboratory second-stage Polaris motor. This test required six-component measurements, and the possibility of a catastrophic malfunction was high.

The general arrangement of motor and test stand is shown in Fig. 21. A view of the forward end of the motor is shown in Fig. 22. Two of the flexure–cells can be seen below the horizontal slab of armor plate to which the motor is attached. The toothed hoop around the center of the motor is an antiflight device. The destruct device, a linear shaped charge, cannot be seen but is located beneath the motor skirt and against the forward bulkhead as a loop that includes the two thrust-reversal ports visible in the photograph. Figure 23 is a view of the after end of the motor; three of the lower flexure–cells can be seen. A fourth flexure–cell, for side-thrust measurement, can be seen in the center of the photograph. The two triangular side supports serve to resist yaw forces.

The only force that was not measured by means of a flexure–cell was axial thrust. A conventional Baldwin strain-gage load cell was used for this measurement. The cell was wrapped in asbestos for protection and was supported by a rod flexure at each end.

The results of the detonation of the destruct device, followed by the firing of the propellant, can be seen in Fig. 24 and 25. All flexure–cells were undamaged. As can be seen in Fig. 24, the rod flexure attached to the Baldwin cell was bent. This is believed to have been due to heat from the large opening in the forward end of the motor.

No attempt is made here to give details of the destruct test. It is desired only to show that the stand was structurally sound following the test, with the exception of the one bent rod flexure.

## CONCLUSIONS AND RECOMMENDATIONS

The use of flexure–cells in rocket-motor testing seems to be promising, especially in tests requiring thrust measurement of fairly low accuracy and in which destruction of the test stand and fixtures is probable. These items are lower in cost than commercial flexure pivots and load cells.

A more extensive test and evaluation program should be conducted on the flexure–cell. It may prove that the first set of flexure–cells was overdesigned. If such is the case, the stiffness of the flexure–cells can be reduced by decreasing the diameter of the column.

It is also suggested that the surface of the flexure–cell be finished to a roughness of 60 μin or less. The first flexures manufactured did not have a fine finish in the region where strain gages were applied. As a result of the rough surface, difficulty was encountered in applying the gages and an electrical imbalance was found in the gage circuit.

A more precise value of *E*, the modulus of elasticity, should be determined for the heat-treated 4130 steel.

With the first flexure–cells, a length of 12 inches was assumed and used. It would be of value to determine exactly which lengths are most suitable for certain load conditions. It may prove that a longer length than 12 inches is possible. This would reduce the stiffness of the flexure–cell and make it even more promising.

For future tests using the flexure–cell, the test stand could be simplified by reducing from four to three the number of flexure–cells holding the mounting plate, with the following advantages: there would be one less flexure–cell to fabricate, and since three points define a plane, the setup of the test stand would be simplified, especially in case of irregularities in the main base plate.

The proposed arrangement of the three flexure–cells is two flexure–cells in a plane through the center of gravity of the motor at right angles to the motor axis, and one flexure–cell aft of the motor center of gravity.

## Appendix A
# THEORETICAL RESTRAINT TO TEST-STAND MOTION CONTRIBUTED BY EACH FLEXURE–CELL

In deriving the equations for the theoretical restraint to test-stand motion provided by the individual flexure–cells and by the column flexure–load cell assembly, a completely rigorous mathematical derivation was not followed because of the complications that would have resulted. The results should be reasonably accurate, none the less.

## NOMENCLATURE

$E$ = Modulus of elasticity of the flexure–cells and the column flexures, assumed to be 30,000,000 psi

$I$ = Moment of inertia of flexure–cell and column flexure, assumed constant for the former

$L$ = Length of member

$M_A$ = Bending moment at end A, $M_A = PL - M_B$

$M_B$ = Bending moment at end B

$M(x)$ = Bending moment at any distance $x$ from end A

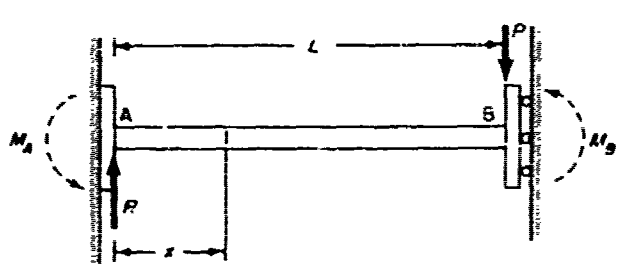
$P$ = Force required for deflection of end B

$R$ = Wall reaction ($R = P$)

$y$ = Deflection perpendicular to axis of piece: in final equations $y$ is the deflection, or offset, at end B

## BOTTOM AND SIDE FLEXURE–CELLS

In the bottom and side flexure–cells the end faces remain parallel to one another.



$$M(x) = Rx - M_A - Px - PL + M_B$$

$$\frac{d^2y}{dx^2} = \frac{1}{EI}(Px - PL + M_B)$$

$$dy/dx = 1/EI[(Px^2/2) - PLx + M_B x + C_1]$$

$$y = 1/EI[(Px^3/6) - (PLx^2/2) + (M_B x^2/2) + C_1 x + C_2]$$

For the boundary conditions

$x = 0$:

$$dy/dx = 0, \text{ hence } C_1 = 0$$

$$y = 0, \text{ hence } C_2 = 0$$

$x = L$:

$$dy/dx = 0, \text{ hence } M_B = PL/2$$

Therefore,

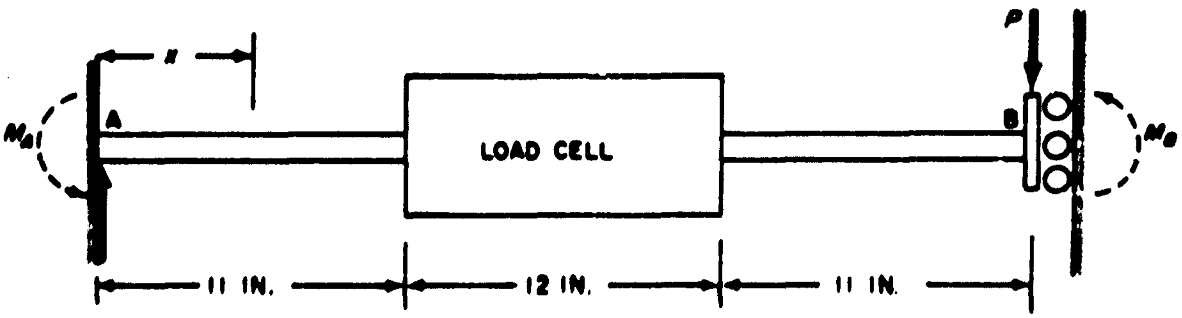
$$y = 1/EI\,[(Px^3/6) - (PLx^2/4)]$$

At $x = L$

$$y = 1/EI\,(-PL^3/12) \qquad \text{or} \qquad P = 12\,EIy/L^3$$

Note: In evaluating this level of restraint per 0.01-inch offset, a mean length of 11 inches was used for $L$, based on the assumption that the moment of inertia, $I$, was constant. The results of the evaluation of the equation are shown in the table on p. 1.

## AXIAL COLUMN FLEXURE–LOAD CELL ASSEMBLY




*REGION* $0 < x < 11$ *INCHES*

$$M(x) = Rx - M_A - Px - M_A$$

$$\frac{d^2y}{dx^2} = \frac{1}{EI}(Px - M_A)$$

$$dy/dx = 1/EI[(Px^2/2) - M_A x] \qquad (1)$$

$$y = 1/EI\,(Px^3/6 - M_A x^2/2) \qquad (2)$$

*REGION 11 INCHES < x < 23 INCHES*

$$M(x) = 0$$

$$\frac{d^2y}{dx^2} = \frac{1}{EI}(0)$$

$$dy/dx = C_3 \qquad (3)$$

$$y = C_3x + C_4 \qquad (4)$$

For the boundary conditions $x = 11$ inches, Eq. (1) = Eq. (3), hence

$$C_3 = (121P/2EI) - (11\,M_A/EI)$$

Also, Eq. 2 = Eq. 4, hence

$$C_4 = (121M_A/2EI) - (1.331P/3EI)$$

Equation (3) becomes

$$dy/dx = (121P/2EI) - (11M_A/EI) \qquad (5)$$

and Eq. 4 becomes

$$y = (121Px/2EI) - (11M_Ax/EI) - (1.331P/3EI) + (121M_A/2EI) \qquad (6)$$

*REGION 23 INCHES < x < 34 INCHES*

$$\frac{d^2y}{dx^2} = \frac{1}{EI}(Px - M_A)$$

$$dy/dx = 1/EI\,[(Px^2/2) - M_Ax + C_5] \qquad (7)$$

$$y = 1/EI\,[(Px^3/6) - (M_Ax^2/2) + C_5x + C_6] \qquad (8)$$

For the boundary condition $x = 23$ inches, Eq. (5) = Eq. (7), hence

$$C_5 = 12M_A - 204P$$

Also, Eq. (6) = Eq. (8), hence

$$C_6 = 3{,}612P - 204M_A$$

For the boundary condition $x = 34$ inches, Eq. (7) = 0, from which

$$M_A = 17P$$

Hence, Eq. (8) becomes

$$y = [(Px^3/6EI) - (17Px^2/2EI) + (144P/EI)] \qquad (9)$$

Also, Eq. (9) yields

$$y = 3{,}131P/EI \qquad \text{or} \qquad P = yEI/3{,}131$$

The results of the evaluation of the equation are shown in the table on p. 1

# Appendix B

## ESTIMATED STEADY-STATE ACCURACY OF THRUST MEASUREMENT USING FLEXURE–CELLS ON SIX-COMPONENT TEST STAND

In determining the accuracy of thrust measurement, thrust readings are assumed for the side flexure–cells, the bottom flexure–cells, and the axial load cell. For each of these recorded loads, the elastic deformation imposed on the particular members is computed using a length of 14 inches to give a maximum value of compressive or tensile deformation. This deformation, then, causes an offset of the same magnitude to exist in the remaining flexure–cells and/or the column flexure–load cell assembly. The offset imposed determines the unmeasured restraint contributed by these other members, from which the approximate accuracy of the original thrust reading can be determined.

### NOMENCLATURE

$P$ = Thrust measurement recorded

$L$ = Length of member undergoing deformation caused by the imposed thrust, taken as 14 inches.

$A$ = Cross-sectional area of member

$E$ = Modulus of elasticity of steel, taken as 30,000,000 psi

$\delta$ = Elastic deformation caused by imposed load

### STEADY-STATE ACCURACY OF SIDE FLEXURE–CELL THRUST READING

If the two side flexure–cells each read 1,000 pounds of thrust, then

$$\delta(\text{side}) = PL/AE = (1{,}000)(14)/(0.282)(30 \times 10^6) = 16.55 \times 10^{-4} \text{ in.}$$

With bottom flexure–cells having an offset of $16.55 \times 10^{-4}$ inch their total restraint is

$$(4)(42)(0.001655)/0.01 = 27.8 \text{ lb}$$

With a column flexure and load-cell assembly having an offset of $1.655 \times 10^{-3}$ inch, the total restraint is

$$(4.7)(0.001655)/0.01 = 0.8 \text{ lb}$$

Hence, total restraining, unmeasured force is 28.6 pounds and the percentage accuracy of the original 2,000-pound thrust reading is 1.43 percent on side-thrust measurements.

### STEADY-STATE ACCURACY OF BOTTOM FLEXURE–CELL THRUST READING

If the four bottom flexure–cells each read 1,000 pounds of thrust, then

$$\delta(\text{bottom}) = PL/AE = (1{,}000)(14)/(0.442)(30 \times 10^6) = 1.06 \times 10^{-3} \text{ in.}$$

With side flexure—cells having an offset of $1.06 \times 10^{-3}$ inch, their total restraint is

$$(2)(17.2)(0.00106)/0.01 = 3.65 \text{ lb}$$

With column flexures and load cell assembly having an offset of $1.06 \times 10^{-3}$ inch, the total restraint is

$$(4.7)(0.00106)/0.01 = 0.5 \text{ lb}$$

Hence, total restraining, unmeasured force is 4.15 pounds and the percentage accuracy of the original 4,000-pound thrust reading is 0.105 percent on vertical-thrust measurements.

## STEADY-STATE ACCURACY OF AXIAL LOAD-CELL THRUST READING

If the axial load cell reads 50,000 pounds of thrust, then

$$\delta\,(\text{load cell}) = 0.01 \text{ in.}$$

$$\delta\,(\text{column flexures}) = (2)(50{,}000)(14)/(0.785)(30 \times 10^{-6}) = 0.06 \text{ in.}$$

Hence,

$$\delta\,(\text{total}) = 0.07 \text{ in.}$$

With bottom flexure—cells having an offset of 0.07 inch, the total restraint is

$$(4)(42)(0.07)/0.01 = 1.175 \text{ lb}$$

With side flexure—cells having an offset of 0.07 inch, the total restraint is

$$(2)(17.2)(0.07)/0.01 = 240 \text{ lb}$$

Hence, the total restraining, unmeasured force is 1,415 pounds, and the percentage accuracy of the original 50,000-pound thrust reading is 2.83 percent on axial thrust measurements.

It should be noted that these restraints are highly repeatable and could easily be accounted for with in-place calibration of the test stand.

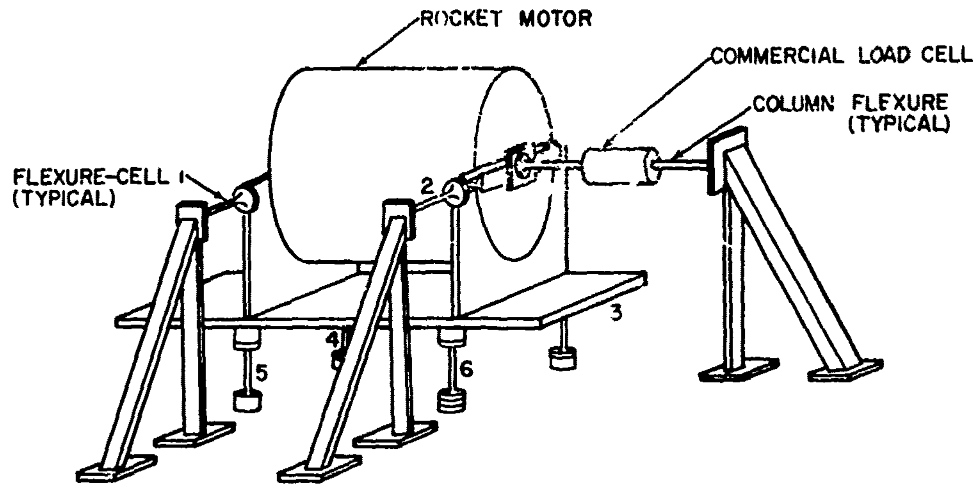



FIG. 1. Six-Component Test Stand Using Six Flexure-Cells, Two Column Flexures, and One 50,000-Pound Universal Load Cell.

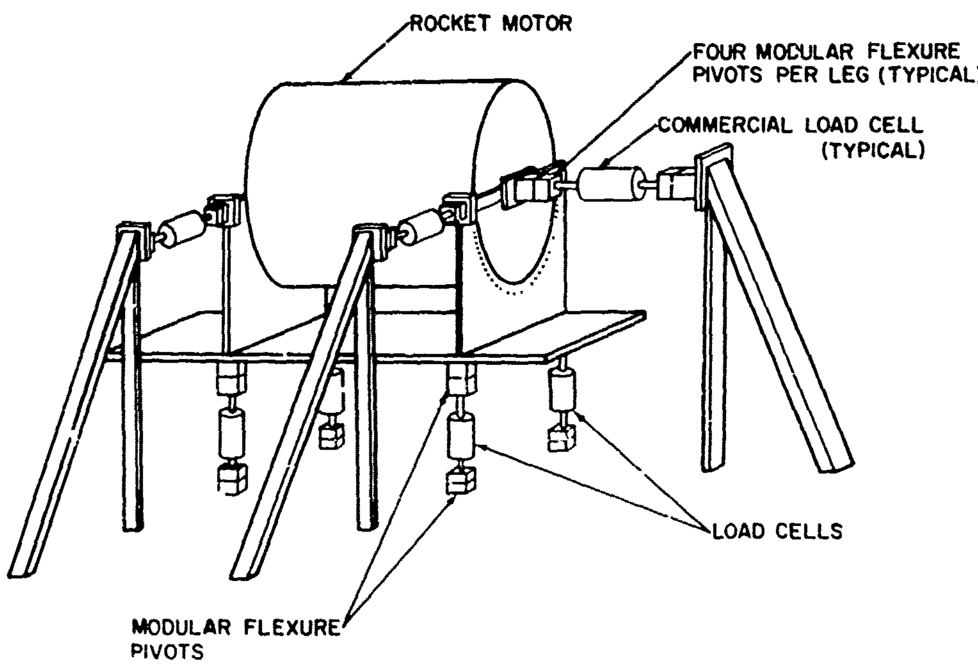
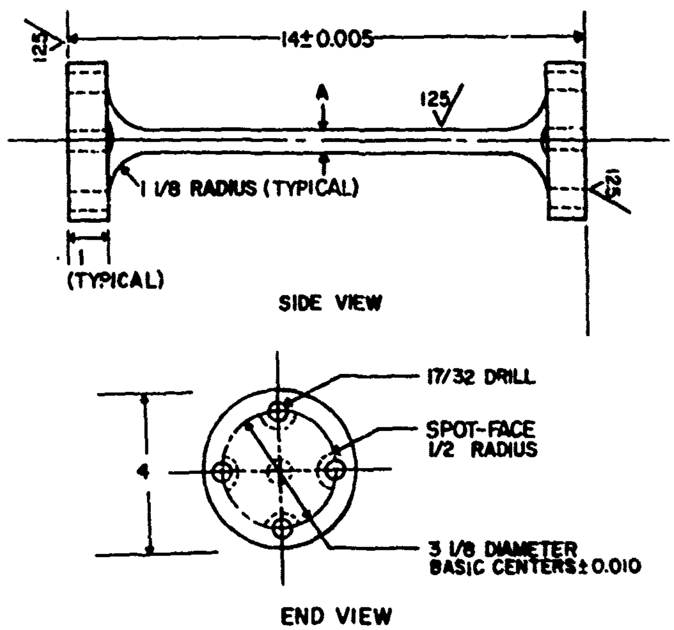



FIG. 2. Six-Component Test Stand Using Twenty-Eight Commercial Flexure Pivots and Seven Load Cells.




DIMENSION A

| | | |
|---|---|---|
| SIDE FLEXURE-CELL | $0.600 {}^{+0.010}_{-0.000}$ | (FLEXURE-CELLS 1 AND 2) |
| BOTTOM FLEXURE-CELL | $0.750 {}^{+0.010}_{-0.000}$ | (FLEXURE-CELLS 3 THROUGH 6) |

FIG. 3. Typical Flexure-Cell.

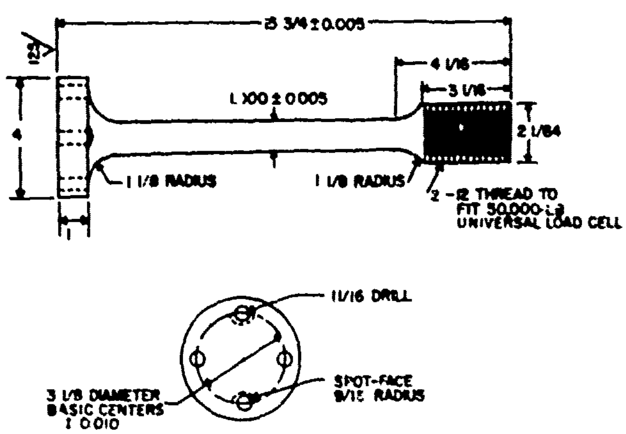



FIG. 4. Axial Column Flexure.

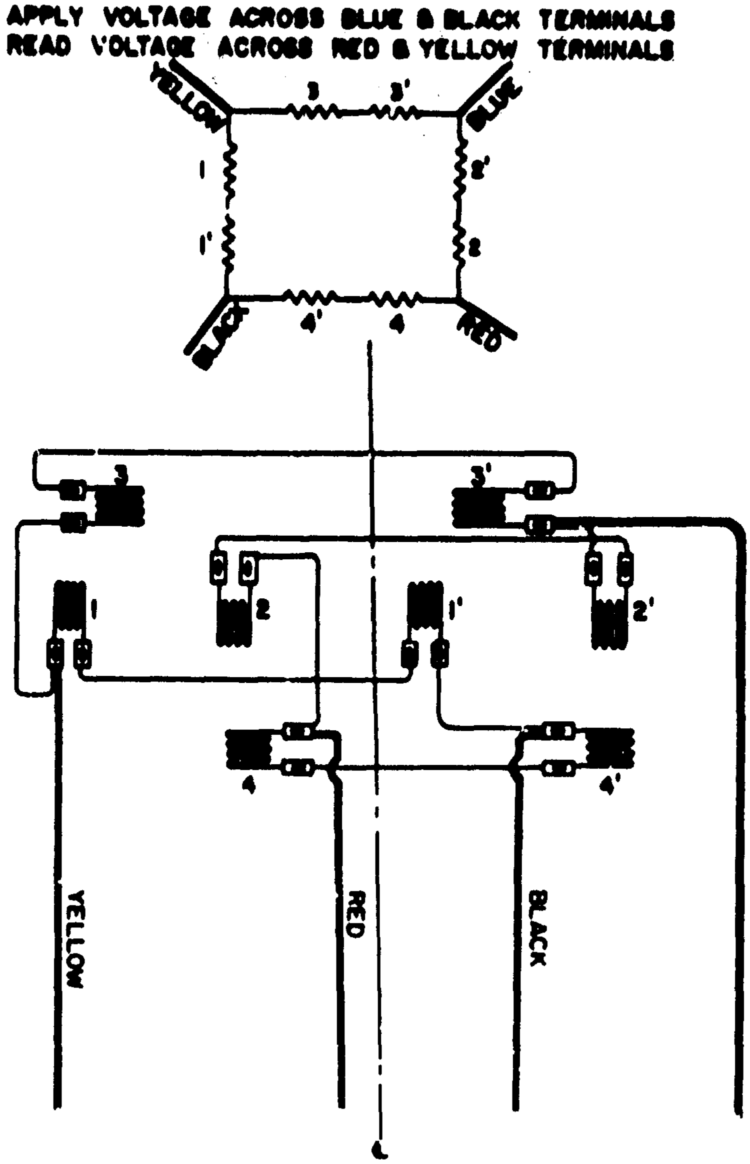



FIG. 5. Strain-Gage Location and Wiring Layout for Flexure-Cell.

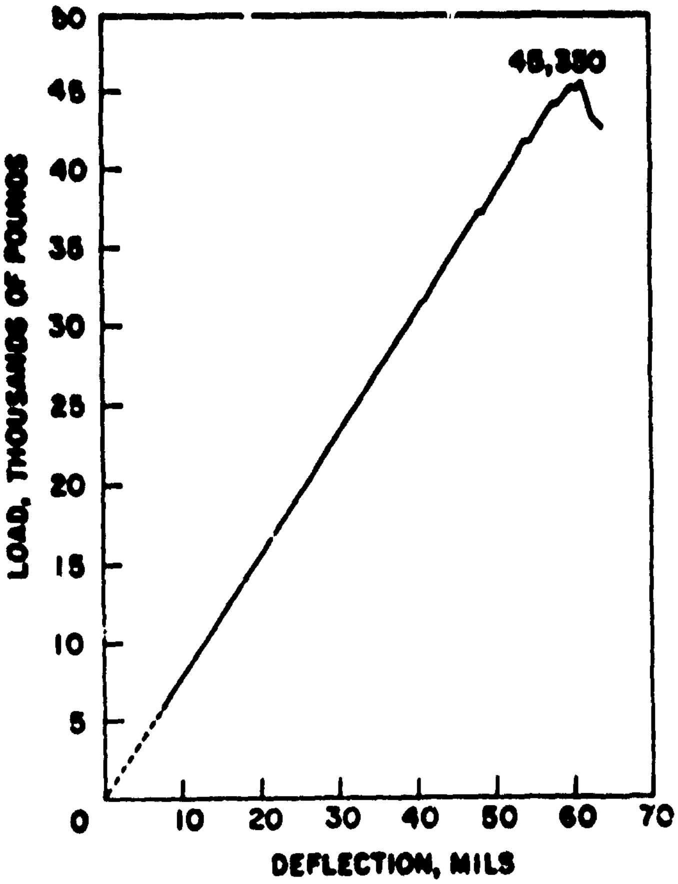



FIG. 6. Applied Load Versus Vertical Deflection Between Flange Faces of 0.60-Inch-Diameter Flexure-Cell With 0.10-Inch Offset.

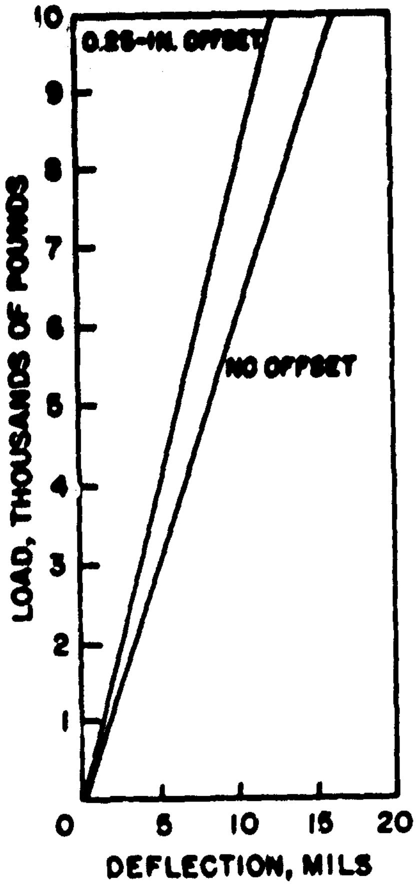



FIG. 7. Applied Load Versus Vertical Deflection Between Flange Faces of 0.60-Inch-Diameter Flexure-Cell.

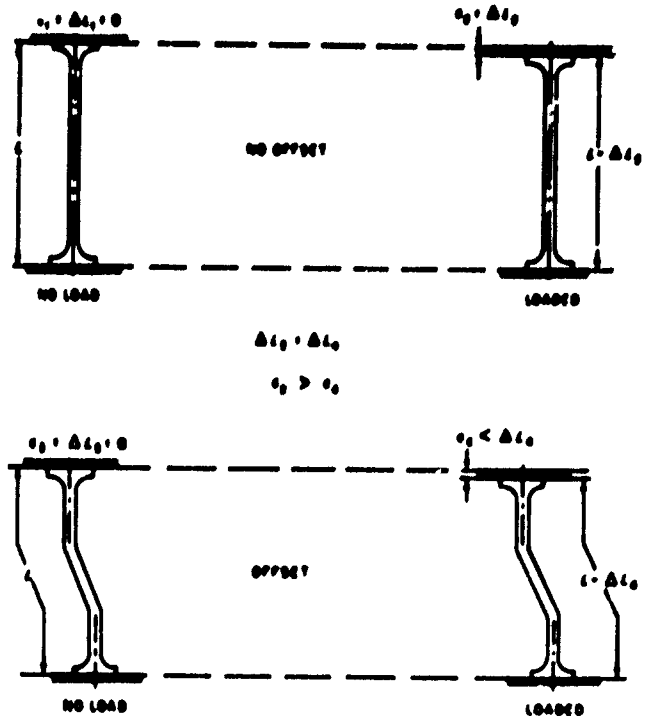



FIG. 8. Flexure-Cell Configuration With and Without Load and Offset. ε Is the Measured Deflection Between Flange Faces.

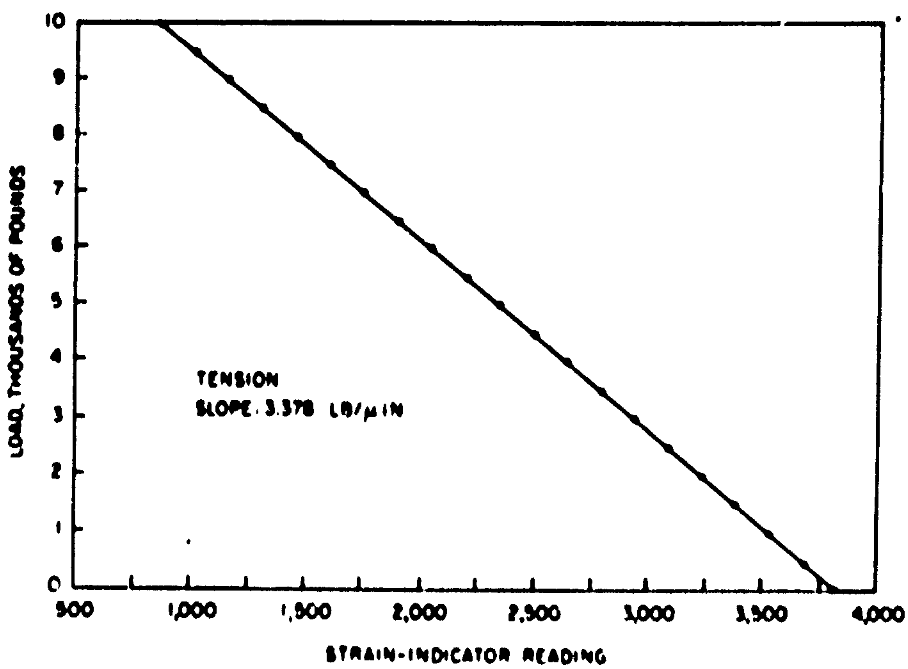



FIG. 9. Load Versus Strain-Indicator Reading for Side Flexure 1, No offset.

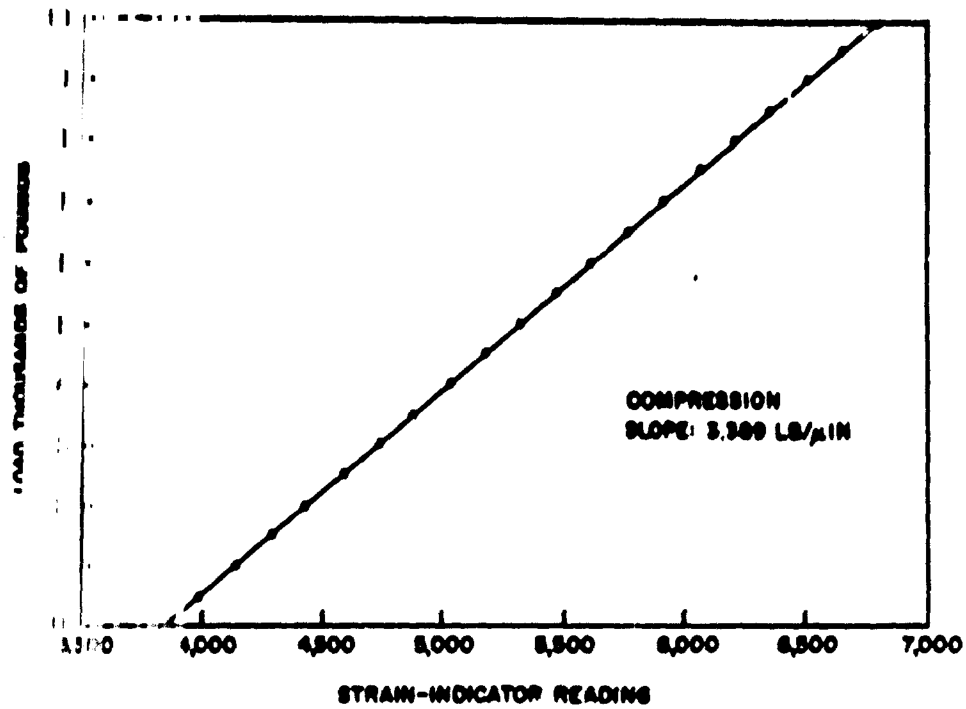



FIG. 10. Load Versus Strain-Indicator Reading for Side Flexure 1. No offset.

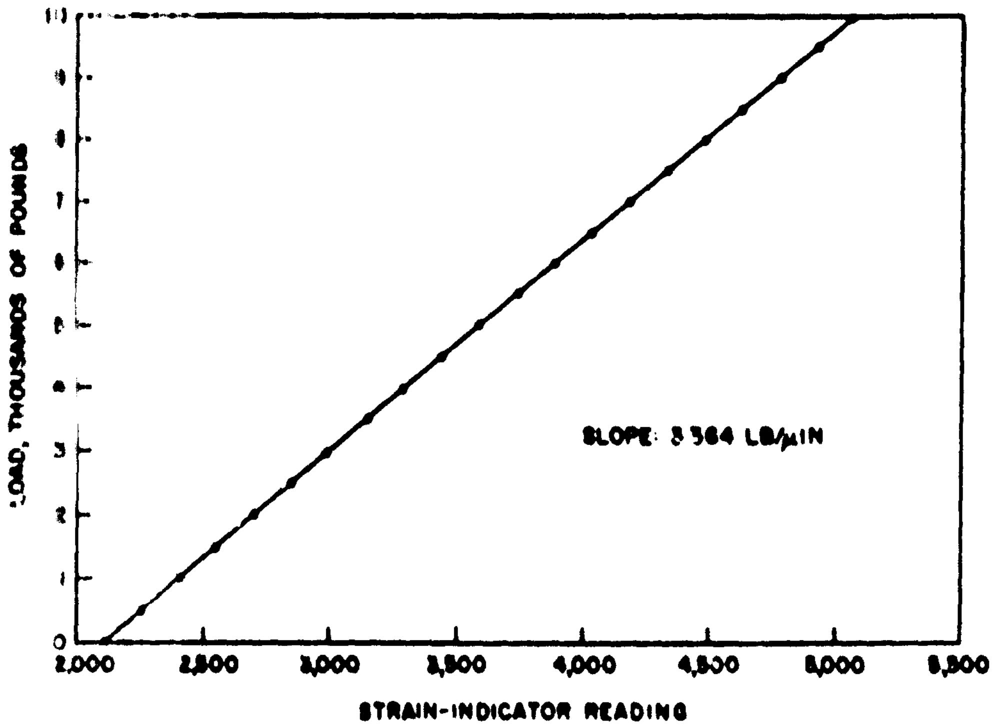



FIG. 11 Load Versus Strain-Indicator Reading for Side Flexure 2, Under Tension.

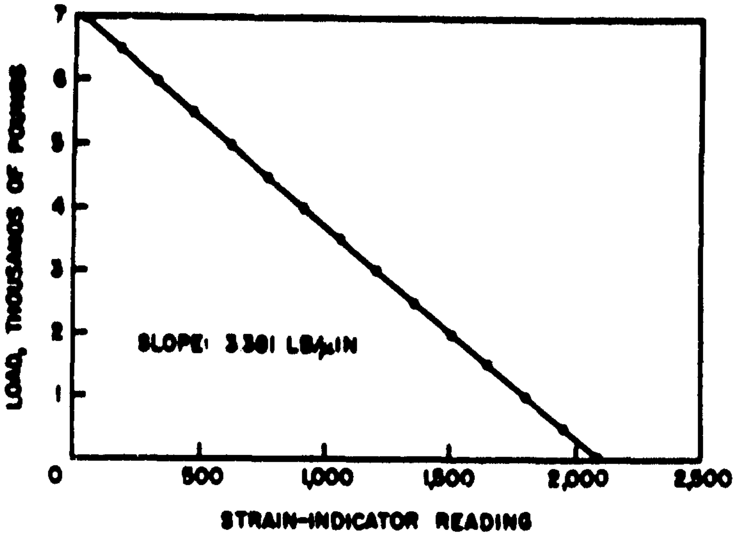



FIG. 12. Load Versus Strain-Indicator Reading for Side Flexure 2, Under Compression.

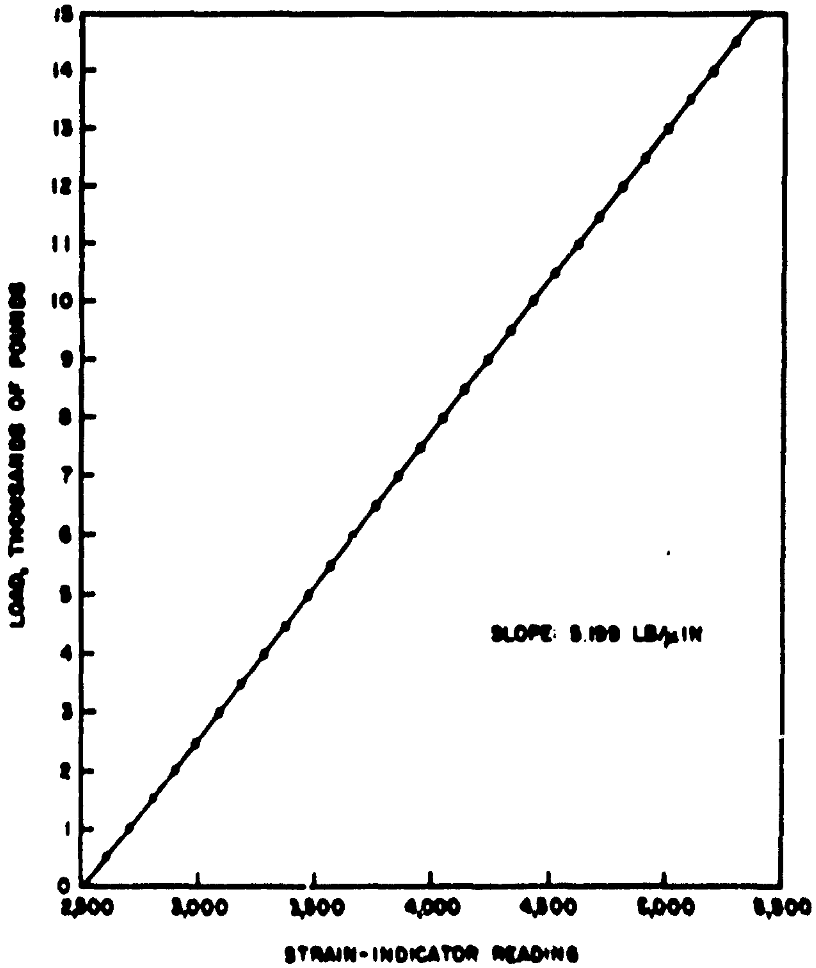



FIG. 13. Load Versus Strain-Indicator Reading for Bottom Flexure 3, Under Tension, No offset.

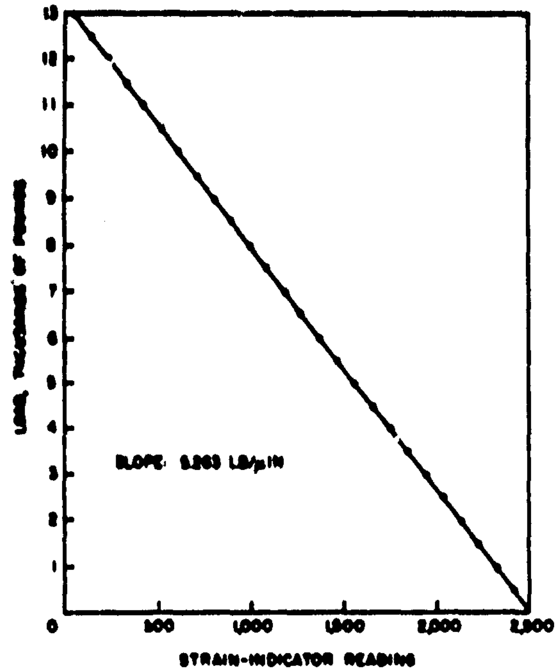



FIG 14. Load Versus Strain-Indicator Reading for Bottom Flexure 3, Under Compression. No offset.

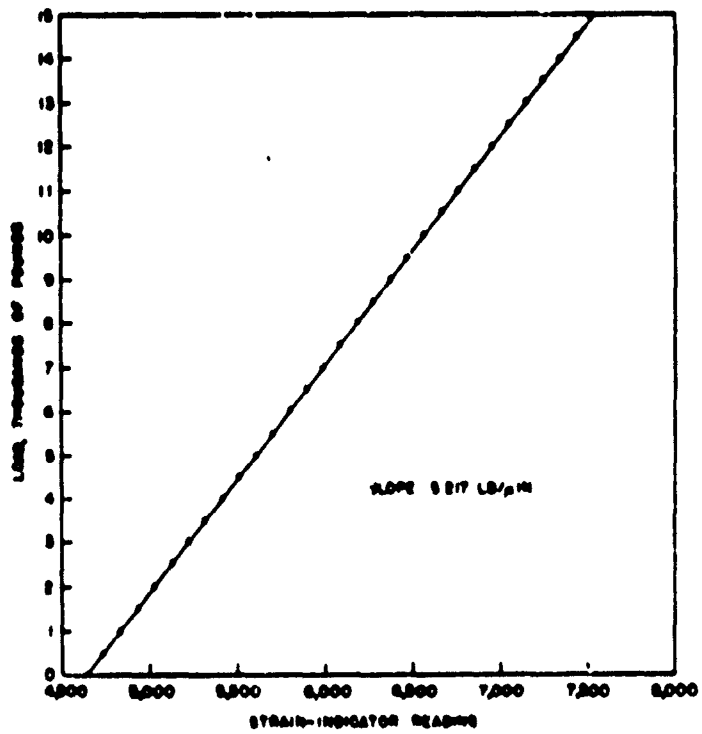



FIG. 15. Load Versus Strain-Indicator Reading for Bottom Flexure 4, Under Tension.

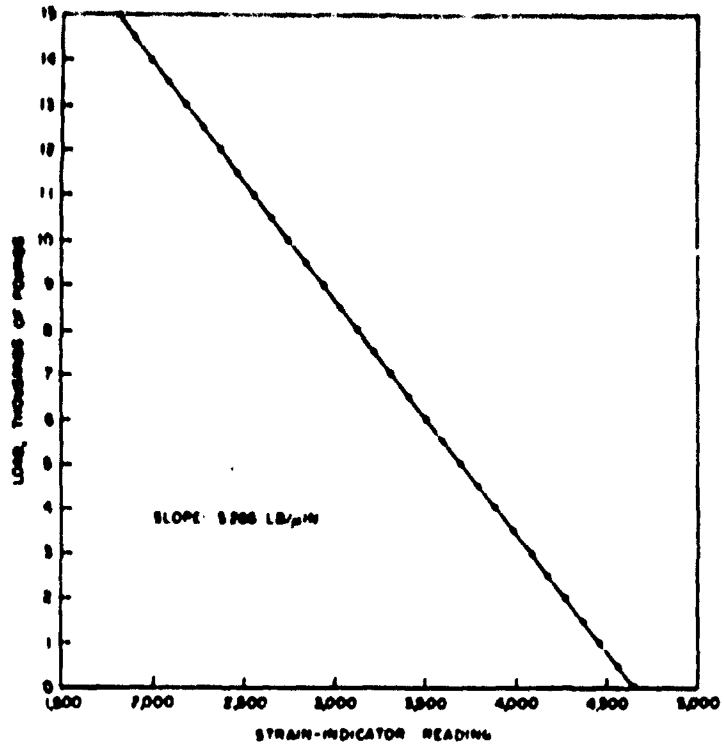



FIG. 16. Load Versus Strain-Indicator Reading for Bottom Flexure 4, Under Compression.

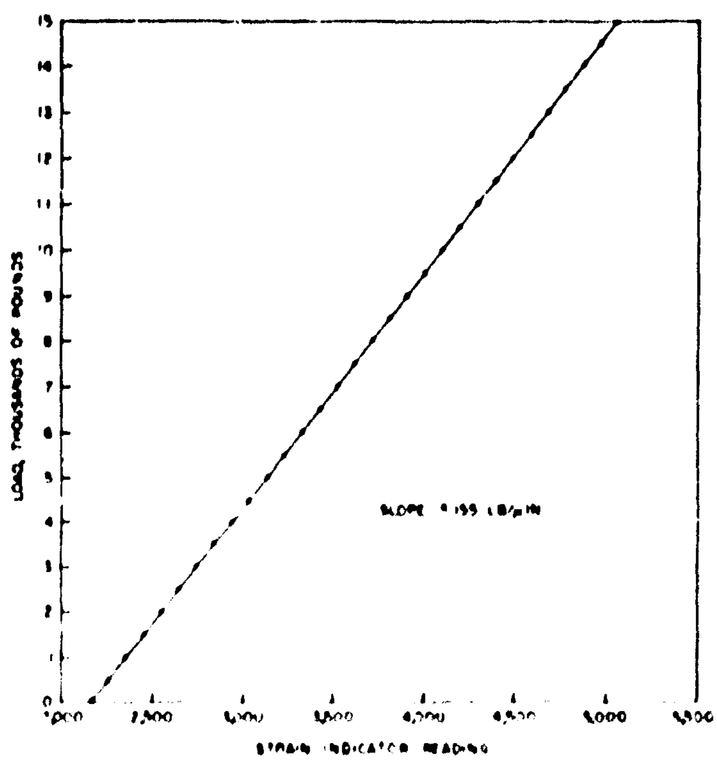



FIG. 17. Load Versus Strain-Indicator Reading for Bottom Flexure 5, Under Tension.

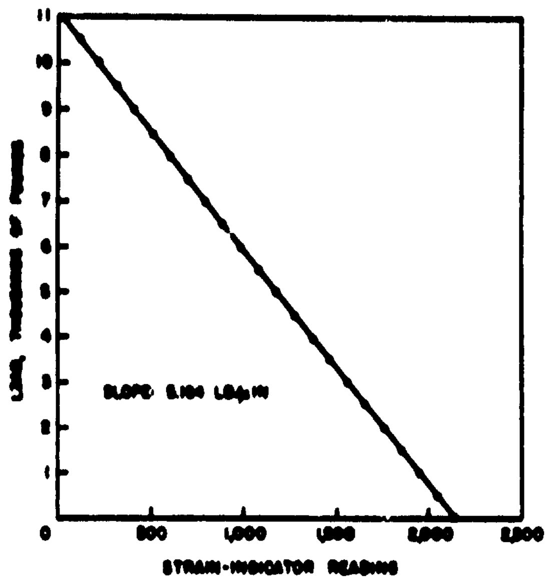



FIG. 18. Load Versus Strain-Indicator Reading for Bottom Flexure 5, Under Compression.

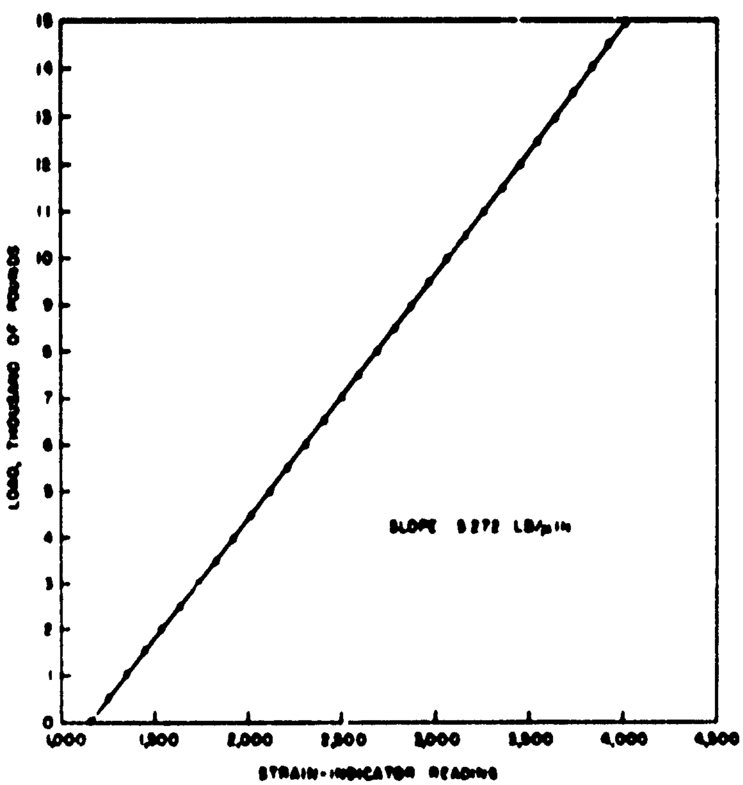



FIG. 19. Load Versus Strain-Indicator Reading for Bottom Flexure 6, Under Tension.

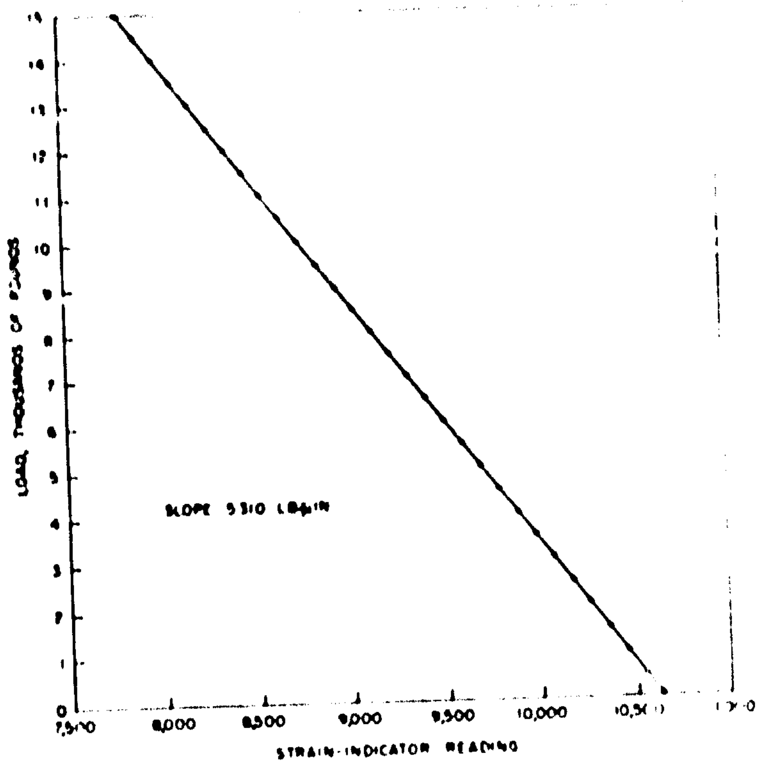



FIG. 20. Load Versus Strain-Indicator Reading for Bottom Flexure 6, Under Compression.

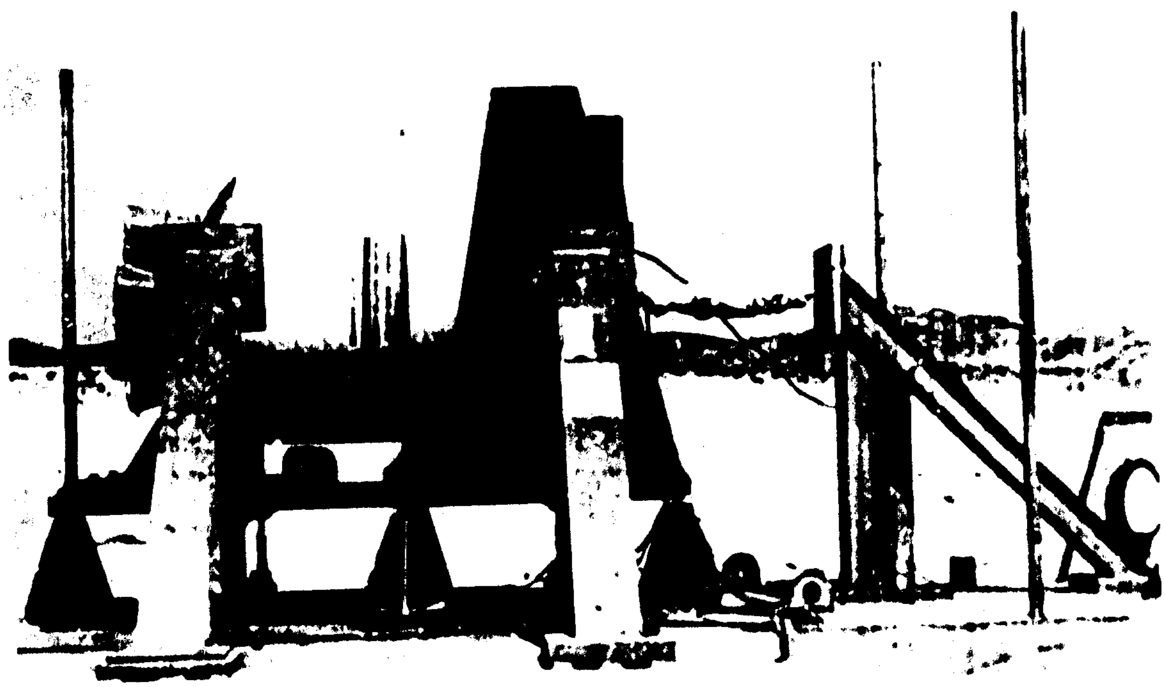


FIG. 21. Six-Component Thrust-M[illegible] and Assembly [illegible]

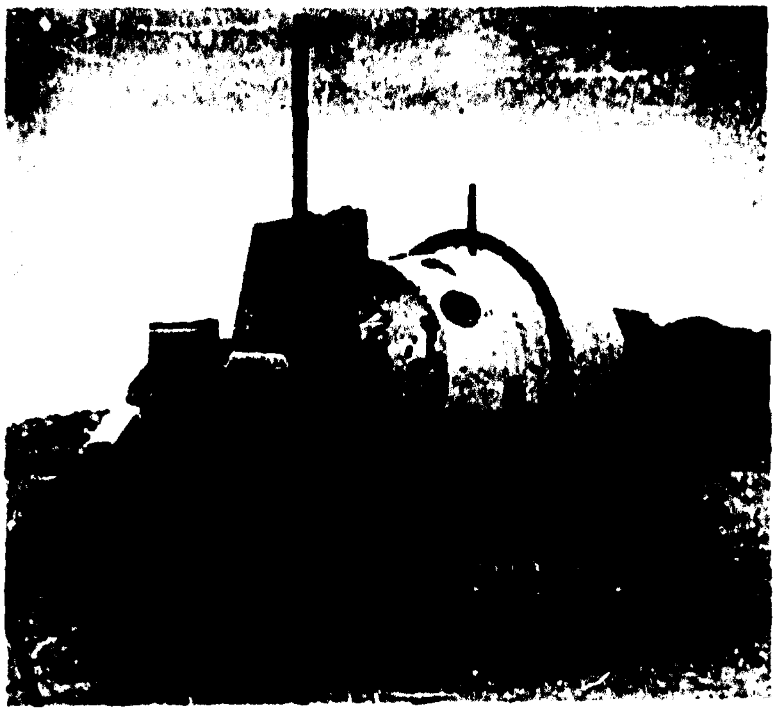


FIG. 22. Forward End of Motor.

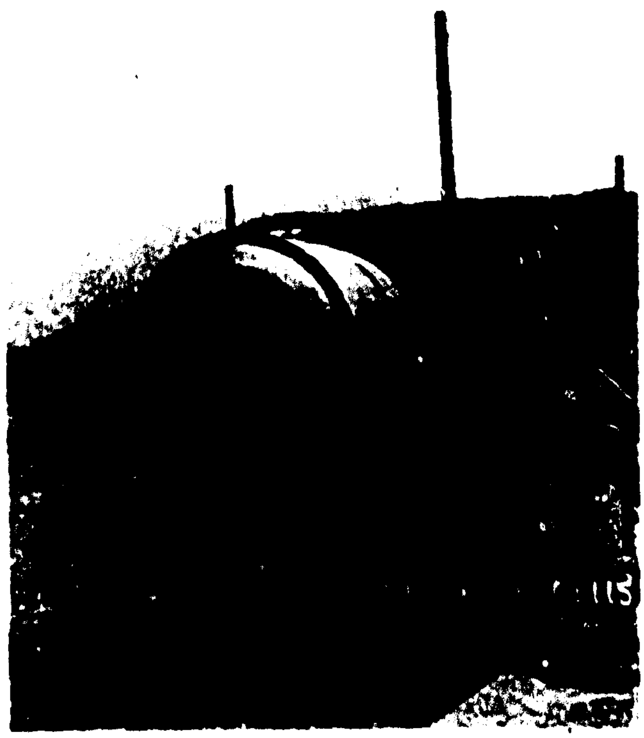


FIG. 23. After End of Motor.

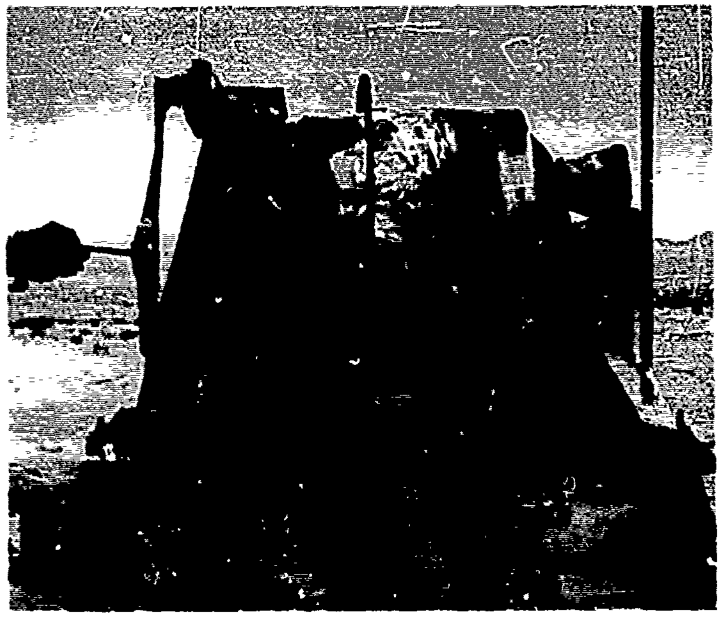


FIG. 24. Post-Firing View of Motor Test-Stand Assembly.

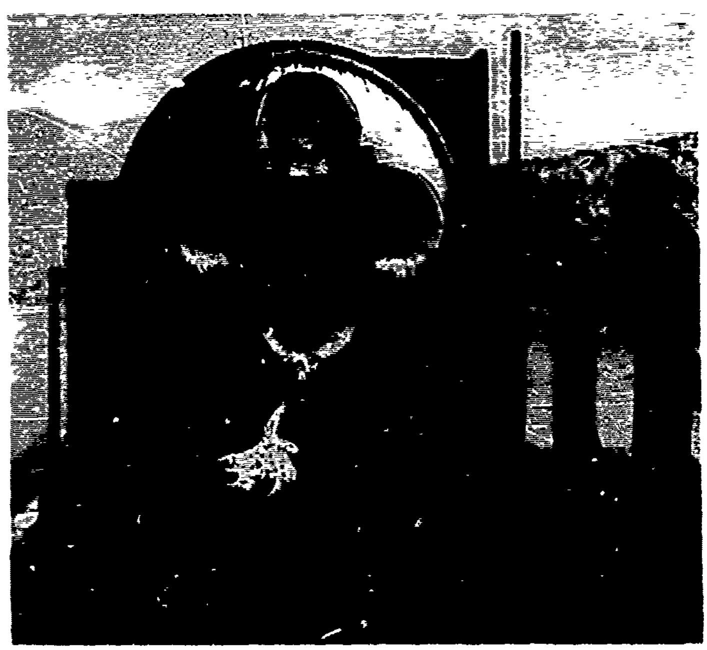


FIG. 25. Post-Firing View of After End of Motor.

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ABSTRACT. The report discusses the design and evaluation of the flexure—cell, a flanged column with strain gages attached. Tables, graphs, and equations show the characteristics of the flexure—cell. Results show that this flexible test-stand mount is cheap, hence expendable, and capable of thrust measurements of fairly high accuracy.

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